LBNL-45080

Performance Diagnostics of Thermal Distribution Systems in Light Commercial Buildings

Tengfang T. Xu, Mark P. Modera, and Remi F. Carrie
Lawrence Berkeley National Laboratory

ABSTRACT

This paper presents findings from field performance testing of five thermal distribution systems in four light commercial buildings. The systems studied are located in Sacramento and Pleasanton, California, and are connected to typical rooftop packaged units serving small office spaces. All of the buildings are single-story office-buildings with floor areas less than 1,000 m^2. The field study included distribution system characterization of the five duct systems, and short-term refrigerant-side monitoring on two of the packaged units. The air leakage results are presented in terms of specific effective leakage areas (ELAs), the ASHRAE-defined duct leakage classes, leakage flowrate (from flow subtraction or derived from ELA and operating pressures), and air leakage ratios (i.e., fractional loss of fan flow). Comparisons are made with results from previous studies on light commercial buildings. The specific ELAs ranged from 0.8 to 5.3 cm^2 per m^2 of floor area served, and the leakage classes ranged from 232 to 414, making these duct systems much leakier than the "unsealed ductwork" classification by ASHRAE. The air leakage ratios were approximately 10% of fan-supplied airflow on average, which was considerably lower than previous studies. Conduction energy losses calculated by temperature measurements in the duct systems were found to be significant, ranging from 9% to 24% of capacity. Refrigerant-side monitoring of temperature and pressure were used to calculate equipment efficiencies. The results show that there were considerable energy losses by air leakage and conduction through ducts, and that the losses varied from system to system. Frequent on-off cooling cycling was common with the systems we tested and additional losses were also found due to improper system control. These diagnostic results suggest that efficiency improvement in thermal distribution could be achieved via better system control, design and sizing, and by duct sealing.

INTRODUCTION

According to the Commercial Buildings Energy Consumption Survey by the Energy Information Agency (EIA 1997), about 650 billion kWh of energy is annually consumed for space conditioning and ventilation in the 4.6 million commercial buildings in the U.S. Commercial buildings with floor areas less than 930 m^2 (10,000 ft^2), termed "light commercial" buildings in this paper, make up approximately three quarters of non-residential buildings in the U.S. and California, while accounting for over 20% of the commercial-building floor area. Given that California commercial buildings have a total floor area of 0.55 billion square meters (6 billion square feet), California light commercial buildings have a total floor area of over 0.11 billion square meters (1.2 billion square feet).

According to the California Energy Commission (CEC 1998), each year California commercial buildings use 35% of statewide electricity consumption, and 16% of statewide gas consumption. About one-third of the electricity and gas used in commercial buildings is for space conditioning, a significant portion of which passes through thermal distribution systems in these buildings. In addition, 8.6 billion kWh per year is consumed by fans and pumps to move the thermal energy through these systems in California commercial buildings. Overall, approximately 12 billion kWh is used for space conditioning and distribution in California light commercial buildings.

The duct systems attached to the rooftop packaged units typically found in light commercial buildings are similar to residential duct systems. Previous field characterizations of light-commercial ducts in California found that air leakage from supply ducts equals approximately one quarter of airflow through system supply-fans (Delp et al. 1998). The field study also suggested that the duct air leakage area per unit floor area served by these systems is typically much higher than that for residential buildings. With annual savings estimates of 1 kWh/ft^2 for light commercial buildings, combined with a distribution of the building stock based upon an extensive stock characterization study and technical penetration estimates (Modera et al. 1999), this translates into California saving potentials of 600 GWh/year and 23 million therms/year.

OBJECTIVES

The objectives of this field study of thermal distribution systems are: 1) to add to the limited existing empirical data on air leakage through ducts in light commercial duct systems; 2) to assess the magnitude of conduction losses (e.g., heat gains through ducts) in light-commercial duct systems; and 3) to assess system performance by monitoring energy use and operation of HVAC equipment.

APPROACH

Our performance characterization of thermal distribution systems involved: 1) physical characterization of duct system by gathering and compiling the physical characteristics of duct systems; 2) measurements of air leakage through duct systems; 3) measurements of operating pressures and air temperatures along duct systems and in their surrounding spaces, and 4) monitoring energy use and refrigerant temperatures of rooftop packaged air conditioners using a field diagnostic tool. We used a tracer gas method to measure total airflow through supply fans. Because the buildings in this study were generally occupied during normal business hours, diagnostic testing was performed in the evenings so as to be non-obtrusive to building occupants. The following describes the measurement techniques used in this study.

Effective leakage area

Duct air leakage is affected by the specific geometry of joints and seams along the duct, the type of sealing material and method, and the operating pressure across the duct work. To characterize the airtightness of thermal distribution systems, the effective leakage areas (ELAs) of isolated sections of duct systems were measured using fan-pressurization procedures. The ELA is defined as the area of a perfect nozzle (i.e., the area of the vena contracta of a sharp-edged orifice) that, at some reference pressure difference, would produce the same flow as that passing through all the leaks in the section being measured. By artificially creating a series of pressure differences across the leaks, the ELA can be determined by fitting the flow and pressure data to Equation (1):

$$Q = \frac{ELA}{10^4}\sqrt{\frac{2\,\Delta P_{\mathrm{ref}}}{\rho}}\left(\frac{\Delta P}{\Delta P_{\mathrm{ref}}}\right)^n , \qquad (1)$$

where Q is the volumetric flow rate ($m^3\ s^{-1}$), ELA is the effective leakage area (cm^2), ΔP is the pressure difference across the leaks in the system (Pa), ΔP_{ref} is a reference pressure difference (Pa), n is the pressure exponent (-), and ρ is the air density ($kg\ m^{-3}$).

The method is well documented in the literature (SMACNA 1985, ASTM 1987, and Delp et al. 1997). We estimate that the true average pressure drop across leaks in the duct may vary by ±2 Pa from the average measured static pressure in the duct during the ELA measurements, which gives an uncertainty of 8% in the average measured pressure across leaks, based on the reference pressure of 25 Pa normally used for characterization of duct leakage. Given the uncertainties in measured air flowrate (±3%) through the fan pressurization equipment, the uncertainty in the measured ELA was estimated to be about ±8% if the pressure exponent *n* is close to 0.6. The reference pressure differential of 25 Pa is based upon field data that shows this to be a common average pressure across duct leaks during normal fan operation in residential and light-commercial building. To allow comparisons between different building systems, duct system ELAs were normalized by the floor area served by the duct system, as well as by the surface area of the ductwork.

Duct leakage class

The leakage class, C_L, is another metric (ASHRAE 1997) used to characterize the leakage rate per unit area of duct surface at a 250 Pa pressure differential across the duct leaks. ASHRAE (1997) lists attainable leakage classes ranging from 3 to 12 for "quality construction and sealing practices," but notes that these attainable leakage classes do not account for leakage at connections of ducts to grilles, diffusers, registers, duct-mounted equipment, or access doors. Leakage class is very similar to duct-area-normalized ELA, differing only in the units and the reference pressure used for the calculation.

Duct system pressure

Operating pressures in ductwork can be very different from one system to another, and almost always vary considerably along the length of any given system. Therefore, to characterize the air leakage flows from duct systems under normal operating conditions by means of the ELA defined in Eq. (1), it is necessary to measure duct system pressures during normal operation. We assumed that static pressures across the ductwork do not vary over time for a constant-speed fan. Pressures were measured at multiple locations in the ductwork (e.g., plenums, branch locations, and terminal registers) using handheld electronic pressure transducers with a 0.1 Pa resolution (Energy Conservatory: Pressure & Fan Flow Gauge, Model DG3, Minneapolis, Minnesota).

A pressure pan measurement method has been proposed to estimate operating pressures in the ductwork in residential building systems (ASHRAE 1999). In this study, we tested this technique. This involved using a digital pressure gauge connected to a sealed register-size pan that was designed to fully block a register during normal system operation. Its key advantage over direct register pressure measurement is that it is more repeatable (Walker et al. 1998). However, the pressure-pan technique more-or-less measures the pressure at the point where the duct being measured branches off at a Wye or Tee, or at the plenum for a radial system.

Airflow through registers

To measure airflow through supply registers more accurately than it is possible with commercially available passive flow hoods, we used an in-house-designed, fan-powered flow hood. During the measurements, air leaving the register passes through a collection hood, then into a duct connected to a variable-speed fan equipped with an integral flow meter. The fan speed was adjusted manually to maintain a low and steady static-pressure difference between the interior of the collection hood and the room. The flow rate was determined with the fan's integral flow meter. We took multi-point measurements above and below the "proxy zero" pressure difference (e.g., 0±0.5 Pa) between the collection hood interior and the room. This enabled us to interpolate to the flow at "zero" pressure difference. Under the circumstance of "zero" pressure difference between the space and a quiescent spot inside the flow hood, we could assume that the flow rate through the register was only marginally affected by the presence of the flow hood, the boundary conditions seen by the register being the same with and without the hood. Note, however, that the minimum pressure drop across the register should be at least 3 Pa to limit the flow measurement uncertainties to 5%.

Air leakage ratio

Air leakage ratio, defined as the air leakage flow rate under normal operating conditions, divided by the total airflow rate through a cross section upstream of the ductwork, is used to characterize the degree of air leakage from supply-duct systems. To estimate the air leakage ratio through supply-duct systems, we measured the total airflow rate through a cross section upstream of the supply-duct systems using a tracer gas method, and measured the air leakage flow rates using the two methods described below.

The two methods used to estimate air leakage flow rates through supply-duct systems were: a) deriving air leakage flow rates from measured ELAs and operating pressures based upon Eq. (1), and b) calculating air leakage flow rates by taking the difference between upstream airflow rate and the sum of supply-register flow rates.

Ideally, deriving air leakage flows (Q) with Eq. (1) requires that the leakage areas of sections of the ductwork that operate at very different pressures be determined separately. However, this level of detailed measurement was not utilized in this study, mainly because the duct pressures tend to vary continuously along the length of the ductwork in light commercial installations. Thus, the pressures monitored at a limited number of locations may not accurately represent the actual pressure distribution across the leaks in the duct systems. The uncertainty in the air leakage ratio can be calculated from the uncertainties in the fan flow and the leakage flow. With proper calibration and operation of instruments, uncertainties in both the tracer gas concentration and the tracer gas injection-rate can be as low as 2% each, and uncertainties due to an imperfect characterization of the well-mixed tracer concentration downstream of the injection point can be 5%. Adding these together, the maximum bias uncertainty is 9%. Adding precision errors of 5% (in quadrature) due to time quantization of sampling, the resulting overall uncertainty in the measured fan airflow rate using a tracer gas was estimated to be about 11% using the current measurement protocol. The uncertainty in the measured ELA and the measured pressure distributions were estimated to be approximately 8% each (see Effective leakage area section above). This analysis suggests an uncertainty in the air leakage ratio of 16%, corresponding to 8%, 8%, and 11% added in quadrature.

The main limitation of the second technique, flow subtraction, is that the expected difference between the upstream flow rate and sum of register flow rates is often comparable in magnitude to the upstream flow and register flow measurement uncertainties. We might expect a 5% uncertainty in the total register flow rate. As calculated above, the overall uncertainty in the measured fan airflow rate using a tracer gas was estimated to be about 11% with the current measurement protocol. The resulting measurement uncertainty in the air-leakage ratio is then approximately 12%, based upon adding 5% and 11% in quadrature.

Thermal losses through conduction

Thermal losses from duct systems result not only from air leakage but also from heat conduction through the duct walls. The assessment of conduction losses, including convection and radiation losses, focused on the analysis of monitored air temperatures in the systems. Thermal measurements were made with stand-alone temperature loggers in the plenum downstream of the cooling/heating coil, in selected supply registers, in the conditioned space, in the ceiling cavity, and in the outside air. The battery-powered temperature loggers with external temperature sensors were HOBO-Pros (Onset Computer Corporation, Pocasset, MA) with 0.03 °C resolution and an accuracy of ±0.2 °C in high-resolution mode. The temperatures measured by multiple collocated temperature loggers show a maximum span of 0.25 °C and a standard deviation of less than 0.1 °C. Delp et al. 1998 evaluated the energy delivery effectiveness of heat transport through ducts in terms of the duct's "cumulative effectiveness," defined as the ratio of the energy delivered at the register to the potential available at the plenum (upstream of conduction losses). Cumulative effectiveness does not include the impact of duct leakage (i.e., it does not include differences in mass flow at the plenum and the registers). By ignoring latent heat effects, the duct's "cumulative effectiveness" equals to the ratio of the sensible heat capacity for heating or cooling delivered at the registers, to the capacity available at the plenum. Based on the assumptions that the airflow through the ductwork is stable over time and space, and that the impact of leakage flow on temperature change is negligible, the equation for the cumulative effectiveness can be simplified by calculating the temperature differential between the register temperature, the plenum temperature and the reference temperature, respectively. The reference temperature is the air temperature of the conditioned space.

Equipment performance monitoring

Characterization of the performance of thermal distribution systems includes characterizing the cycling characteristics of the cooling/heating equipment, monitoring short-term energy consumption, and monitoring maximum electricity demand. The energy monitoring includes using a diagnostics tool (Field Diagnostics: ACRx, Philadelphia, PA) to collect short-term data on electric energy consumption and equipment efficiency for rooftop packaged units in the field during hot summer days. We performed the monitoring on two systems. To obtain the enthalpy changes in the refrigerant, the refrigerant temperature and pressure were monitored before and after the compressor. The coefficient of performance (COP) is defined as the total output of cooling capacity divided by the total input of work in compressing the refrigerant vapor. The following equation illustrates the calculation of unit's COP during steady-state operation:

$$COP = \frac{(h_1 - h_4)}{(h_2 - h_1)C}, \tag{2}$$

where h_1 is the specific enthalpy of refrigerant leaving the evaporator (kJ/kg) or entering the compressor, h_2 is the specific enthalpy of refrigerant leaving the compressor (kJ/kg), h_4 is the specific enthalpy of refrigerant entering the evaporator (kJ/kg), and C is an empirical coefficient to account for heat losses from the compressor (1.095).

RESULTS

The four buildings in this study were all office buildings with total floor areas ranging between 167 and 745 m^2 (1,800 to 8,024 ft^2), with total building cooling capacities ranging from 11 to 65 kW (3 to 18.5 tons). Table 1 summarizes the physical characteristics of the five rooftop-packaged systems studied in these buildings.

Table 1. Physical characteristics of systems

Building/System Information	UNIT(S)	System 1	System 2	System 3	System 4	System 5
Year Built	-	1988	1988	1996	1996	1996
Cooling Capacity of HVAC system tested	*tons*	3	4	5	5	4
	kW	11	14	18	18	14
Floor Area Served by the HVAC System	*ft^2*	1800	2160	1000	1800	1056
	m^2	167	201	93	167	98
Fan Flow	*cfm*	746	1122	1764	1507	1353
	L/s	352	529	832	711	638
Supply Plenum Static Pressure	*In. WC*	0.11	0.06	0.24	0.12	0.09
	Pa	29	14	61	30	23
Duct Insulated? (Y/N/Part)	-	y	y	y	y	y
Supply Duct Surface Area	*ft^2*	225	291	540	360	274
	m^2	21	27	50	33	25
Return Duct Surface Area	*ft^2*	159	182	320	120	209
	m^2	15	17	30	11	19
Floor Area per Supply Register	*ft^2/register*	360	164	200	360	211
	m^2/register	33	15	19	33	20
Longest Supply Duct Run	*ft*	35	33	96	62	42
	m	11	10	29	19	13
Fan Flow/Capacity	*cfm/ton*	249	281	353	301	338
	L/s/kW	33	38	47	40	45
Fan Flow/Floor Area of Section Measured	*cfm/ft^2*	0.4	0.6	1.8	0.8	1.3
	L/s/m^2	2	3	9	4	7
Fan Flow/Supply Duct Surface Area	*cfm/ft^2*	3	3	3	4	5
	L/s/m^2	17	17	17	21	25
# Supply Registers (All rectangular)	-	5	11	5	5	5
# Return Registers (All rectangular)	-	5	6	5	2	4

SUPPLY DUCT EFFECTIVE LEAKAGE AREA (ELA_{25}). The specific ELA_{25} in this study ranged from 0.8 to 5.3 cm^2 per m^2 of floor area served, with an average value of 2.6 cm^2/m^2 and a standard deviation of 1.8 cm^2/m^2. If we assume that the uncertainty in the measured floor area was 10%, combining this with the 8% uncertainty in ELA_{25}, we obtain a 13% uncertainty in the calculated specific ELA_{25}. On this per-unit-floor-area basis, the average specific ELA_{25} in this study was lower than the average of 3.1 cm^2 per m^2 of floor area reported by Delp et al. (1999), while it was close to the result of 2.7 cm^2/m^2 per floor area reported by Cummings et al. (1996). Normalizing by duct surface area, the specific ELA_{25} in our study ranged from 3.7 to 7.5 cm^2 per m^2 of duct surface area, with an average value of 6.1 cm^2 per m^2 of duct surface area and a standard deviation of 1.4 cm^2 per m^2 of duct surface area.

AIR LEAKAGE CLASS. The total leakage class (supply, return, and air handler) of the small systems measured ranged from 232 to 414, averaging 333 with a standard deviation of 70. The mean value was lower than the 447 value reported by Delp et al. (1999), which reports that the total leakage classes ranged from 130 to over 1,300, with a mean of 447 and a standard deviation of 272 based on over 30 light-commercial systems tested over the previous years. The uncertainty in ELA_{25} was 8% and the uncertainty of calculating duct surface area was 10%, resulting in an uncertainty of 13% in the leakage class calculated from these measurements (excluding exponent errors). These values of measured leakage classes are higher than (by almost an order of magnitude) the ASHRAE value of 48 for "unsealed" rectangular metal ducts (ASHRAE 1997). However, the ASHRAE values, specified for different duct types instead of duct systems, neglect leakage at connections of ducts to grilles, diffusers, registers, duct-mounted equipment, or access doors.

OPERATING PRESSURE. The supply-plenum static pressure relative to the conditioned space ranged from 14 to 61 Pa, with an average of 31 Pa. The measured static pressures ranged from 10 to 24 Pa at the furthest downstream supply register. As indicated in Table 2, the average pressures in the supply plenums in our study were about 50% lower than the average found in a previous study on light commercial buildings (Delp et 1999). Because we only studied five systems, the statistical significance of this difference in the mean values is, however, inconclusive.

Table 2. Comparisons of supply and return plenum operating pressures of current study with those in previous studies

Operating pressures (Pa)	Supply duct sections			Return duct sections		
	Mean	Std. Dev.	Total #	Mean	Std. Dev.	Total #
Small Commercial (present study)	31	18	5	-19	12	5
Small Commercial (Delp et al. 1999)	66	36	30	-43	25	30
Residential (Jump et al. 1996)*	44	N/A	N/A	-64	N/A	N/A

* Unreported

AIR LEAKAGE RATIOS. The average air leakage ratio, the ratio of air leakage flow to the total supply airflow, was approximately 10% with a standard deviation of 6%. Even given the uncertainties of 13-16% in air leakage ratio, this is significantly lower than the 26% of fan flow (average value) reported in a previous California study (Delp et al. 1997). However, it is clear that it is a combination of lower leakage levels and lower operating pressures that is creating this result.

HEAT CONDUCTION LOSSES. In cooling mode, heat gains between the outlet of the cooling coils and the supply registers usually caused supply-air temperatures to increase toward the end of the supply run, thus lowering the cumulative effectiveness of cooling. Table 3 shows the register temperature rises from supply plenum, and the cumulative effectiveness (in parenthesis). The system-average temperature rises between the outlet of the cooling coils and the supply registers due to heat gains ranged from 1.2 to 2.4 °C. The overall cumulative effectiveness $[(T_{register} - T_{room})/(T_{plenum} - T_{room})]$ for light-commercial building systems ranged from 0.76 to 0.91 on average.

Since the weather was mild during the period when the field tests were performed, most of the systems were not operating at their full capacity all the time. One would therefore expect frequent "on-off" cooling operation. In fact, the fractional on-time for cooling cycles in these buildings ranged between 14% and 48% during occupied hours. Not surprisingly, when the cooling-on-time fractions rose, the effectiveness increased. This is caused by the impact of thermal cycling on storage of thermal energy in the ducts, and the resulting temporal variations in duct register temperatures. When the cycle on-time fraction is increased, the energy stored in, and ultimately lost from, the duct system becomes a smaller fraction of the total energy delivered by the system.

Table 3. Cycle-on time fraction, register temperature rise and cumulative effectiveness at registers

System	Fraction of cooling ON-Time	Temperature rise at end of cooling-ON swings (°C)					
		Supply register A	Supply register B	Supply register C	Supply register D	Supply register E	Average
S1 Roof-top Unit	48%	1.1 (0.89)	1.9 (0.80)	-	-	-	1.5 (0.85)
S3 Roof-top Unit	14%	4.1 (0.66)	2.3 (0.73)	1.8 (0.84)	1.8 (0.81)	2.0 (0.79)	2.4 (0.76)
S5 Roof-top Unit	39%	0.9 (0.93)	0.8 (0.94)	0.9 (0.93)	1.4 (0.89)	2.0 (0.85)	1.2 (0.91)

EQUIPMENT PERFORMANCE MONITORING. To assess the performance of the rooftop package units themselves, we monitored the temperatures and pressures of the refrigerant at different locations (e.g., before and after the compressor). Under a steady-state assumption, the coefficient of performance (COP) can be calculated based on the calculated enthalpies of the refrigerant, avoiding the need to measure the electricity energy input to the equipment and the cooling energy output from the evaporator. However, the values of calculated COPs based on instantaneous temperature and pressure data do not necessarily represent the true energy efficiencies of the units. Instantaneous results are compromised by unstable refrigerant flow rates at the beginning of unit start-up, the minimal time needed for the refrigerant tubes to reach thermal balance with ambient conditions and their temperature sensors.

Two systems, System S3 and System S5, were tested in the summer of 1999. From short-term monitoring of one unit's electricity use, we observed that on average the unit was on for 20 to 40 minutes, followed by 5 to 10 minutes off. Figure 1 shows the trend of calculated COP, outside air temperature, space temperature and the electricity energy demand during one day of operation of System S3 in August. From this plot, we observe that the same unit was in continuous operation most of the time (9:00 AM – 6:00 PM). While the temperatures, pressures and flow of the refrigerant in the AC unit may change slightly over time due to variations in the ambient conditions, the instantaneous values should be meaningful in this instance. In this case, COP values over time can be used to assess the unit's operating efficiencies. As expected, the

COP changed with ambient temperature (under constant space temperature conditions).

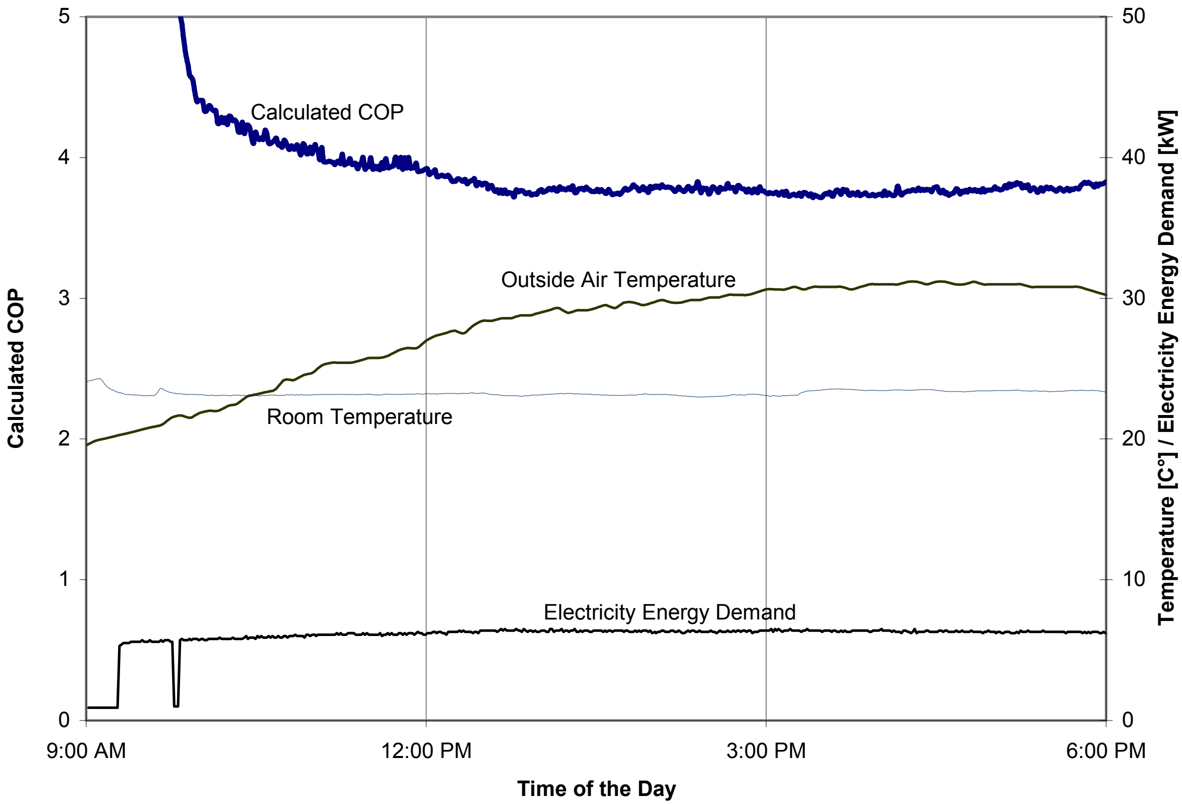

Figure 1. System S3: Calculated COP, outside air temperature, space temperature and electricity energy demand during continuous operation

It should be noted that the data in Figure 1 is an exception rather than the rule. On four of five weekdays in late August, System S3 experienced intermittent operating patterns. Since there are other units serving the same office building as S3, and each system's operation was largely affected by the individual thermostat set point, we cannot at this stage judge the appropriateness of the sizing of unit S3.

Table 4 shows the maximum-hourly and average electricity demand and short-term electricity consumption of systems S3 and S5. From the table we see that the load factors (the ratio of average to maximum hourly demand) for these weeks vary between 18% and 30%, demonstrating the general unattractiveness of light-commercial cooling loads from the point of view of a utility. From the energy use data we find that about 70% of the total electricity use of System S3 occurred between 9 AM and 6 PM, indicating that a considerable portion of energy use occurred outside of normal business hours. Based upon our observations that the S3 office was only open during normal office hours, the ACRx monitoring demonstrates that excessive energy was used to condition spaces during unoccupied periods, indicating control-schedule problems. For System S5, the electricity use of the AC system between 9 AM and 6 PM accounted for 90% of the unit's total electricity use.

Table 4 Short-term electricity energy demand and electricity use

AC System	System S3		System S5	
	Electricity demand (kW)*	One-week electricity use (kWh)	Electricity demand (kW)	One-week electricity use (kWh)
Weekly Average	2.1	353 (70%)**	0.83	140 (90%)*
Maximum Hourly	7.0		4.7	

* Includes fan power and power for controls.

** Percentage in parenthesis represents the ratio of electricity energy use of the AC unit from 9 AM to 6 PM to the total electricity use of the AC unit during the week.

DISCUSSION

The field data presented here confirm the trends reported in Delp et al. 1999 regarding the thermal performance of duct systems in light commercial buildings. Although some of the diagnostic methods and devices have been improved to produce better accuracy and simplicity, further research is still needed to develop complete duct system diagnostic protocols that could be used on a wider scale. For example, it requires significant time to set up, calibrate, and perform the fan-flow measurements using the tracer gas (TG) method. The TG method is sensitive and expensive, and its performance requires expertise. For practical field diagnostics, we need to have a more-simplified protocol that provides comparable accuracy.

Concerning the quoted measurements of duct performance, it should be noted that the thermal load in light-commercial buildings is sensitive to climate, and thus the additional equipment cycling during those periods would cause the cumulative cooling effectiveness to be worse, resulting in lower system efficiencies.

The ACRx field diagnostics tool provides a useful way to monitor an air conditioner's operating performance. The data obtained can be valuable in the following ways: 1) detecting unit's on-and-off operating patterns, which may be used as an indication of system's state of repair, failure, oversizing, or improper control, 2) providing COP performance data during steady-state operation, 3) providing the data of energy use and energy demand of the unit during a certain period of time as selected by users, and 4) continuous collection of data according to users' needs once the tools is setup and working properly. The shortcomings of the tool include its inability to accurately monitor the unit's COP performance during non-steady-state operation. Also, the time required for measurement setup was somewhat long, and telecommunication technique utilized was sometimes unstable.

CONCLUSIONS AND RECOMMENDATIONS

The field characterization provides new data for our understanding of duct system performance. First, there is duct air leakage in light commercial buildings with the average leakage ratio around 10%. There are large variations in the leakage levels across building systems tested, a phenomenon similar to the duct air leakage found in residences (Modera et al. 1989, Yuill et al. 1997) and light commercial buildings (Delp et al. 1999). In addition, duct-systems' thermal conduction losses (including convection and radiation losses) in light commercial buildings were also found to be significant. The supply-air temperature changes associated with these losses ranged from 1.2 °C up to 2.4 °C on average, well above the "designer's rule of thumb" of 0.6°C (1 °F). The thermal losses induced by heat conduction through duct walls were considerable, ranging from approximately one-tenth to a quarter of the cooling capacity from cooling coils. Field data also demonstrated that the cycling of these systems leads to lower duct and therefore system efficiencies.

More field characterization is needed to improve our knowledge on the duct system performance, including energy performance of the HVAC equipment in light commercial buildings. Diagnostic tools need to be improved to provide quick, accurate and complete diagnoses of system performance.

REFERENCES

ASHRAE. 1997. *ASHRAE Handbook of Fundamentals, Chapter 32, "Duct Design."* American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., Atlanta, GA.

ASHRAE. 1999. *Proposed Standard 152P, "Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Systems,"* Draft for Public Review. American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., Atlanta, GA. January 1999.

ASTM. 1987. *"ASTM E-779-87, Standard test method for determining air leakage rate by fan pressurization,"* American Society for Testing and Materials, Philadelphia, PA.

California Energy Commission (CEC). 1998. *"1998 Baseline Energy Outlook,"* P300-98-012, August 1998.

Cummings, J. B., Withers, C. R., Moyer, N., Fairey, P., and McKendry, B. (1996) "Uncontrolled airflow in non-residential buildings," *Florida Solar Energy Center, FSEC-CR-878-96*.

Delp, W.W., J. McWilliams, D. Dickerhoff, D. Wang, and M. Modera. 1999. "Thermal Distribution Losses in Light Commercial Buildings." Submitted to *Energy and Buildings,* Lawrence Berkeley National Laboratory, Berkeley, California.

Delp, W.W., N. Matson, E. Tschudy, and M. Modera. 1998. "Field Investigation of Duct System Performance in California Small commercial Buildings (Round II)" *Proceedings of ACEEE Summer Study 1998. 3: 105-116,* American Council for an Energy Efficient Economy, Washington, D.C.

Delp, W.W., N. Matson, E. Tschudy, M. Modera, and R. Diamond. 1997. "Field investigation of duct system performance in California light-commercial buildings," *LBNL-40102, Lawrence Berkeley National Laboratory,* Berkeley, California.

Energy Information Administration (EIA). 1997. *"Commercial Buildings Characteristics 1995,"* U.S. Department of Energy, August 1997.

Modera, M. P., 1989. "Residential duct system leakage - magnitude, impacts and potential for reduction." *ASHRAE Transactions.* 1989, Vol.95, Part 2, pp. 561-569.

Modera, M., T. Xu, H. Feustel, N. Matson, C. Huizenga, F. Bauman, E. Arens, and T. Borgers, 1999 (revision). "Efficient Thermal Energy Distribution in Commercial Buildings." *LBNL-41365, Lawrence Berkeley National Laboratory,* Berkeley, California. August 1999.

SMACNA. 1985. *HVAC Air Duct Leakage Test Manual.* Chantilly, VA. Sheet Metal and Air Conditioning Contractors' National Association, Inc.

Walker, I., M. Sherman, M. Modera, and J. Siegel. 1998. "Leakage diagnostics, sealant longevity, sizing and technology transfer in residential thermal distribution systems," *LBNL-41118, Lawrence Berkeley National Laboratory,* Berkeley, California.

Xu, T., O. Bechu, F. R. Carrié, D. J. Dickerhoff, W. J. Fisk, E. Franconi, Ø. Kristiansen, R. Levinson, J. McWilliams, D. Wang, and M. P. Modera. 1999. "Commercial Thermal Distribution Systems," *LBNL-44320, Lawrence Berkeley National Laboratory, Berkeley*, California. Final Report to California Energy Commission (CEC) and California Institute for Energy Efficiency (CIEE).

Yuill, G. K, Musser A., 1997. "*Evaluation of residential duct-sealing effectiveness.*" ASHRAE Transactions. 1997, Vol.103, Part 2, pp. 264-271.